Dual-mode switched control of an electropneumatic clutch actuator with input restrictions

Hege Langjord, Tor Arne Johansen* and Christian Bratli†

Abstract— **This paper propose a stabilizing switched controller for an electropneumatic clutch actuator system with input restrictions. These restrictions arise from the fact that the allocation of air to and from the actuator valve is governed by on/off valves. The derived dual-mode controller is a combination of two previous developed switched controllers for the same system, and asymptotic stability in the region of operation of the clutch is proved. The stability property is verified and the controller's performance is investigated through experimental results.**

I. Introduction

The system considered in this paper is an electropneumatic clutch actuator for heavy-duty trucks. This clutch system is well suited for Automated Manual Transmissions (AMT) systems and clutch by wire. AMTs consist of an automated actuated clutch during gear shifts and a manual transmission through the clutch disc, and are an alternative in between Manual Transmissions (MT) systems and Automatic Transmissions (AT) systems. AMTs are the preferred choice in heavy-duty trucks, because AT systems for such high torque transfer are expensive and have a large power loss [1]. Additional advantages of the ATM systems are improved fuel consumption, higher efficiency and reduced clutch wear, hence lower costs, compared to MT systems. Automatic control of the clutch engagement is important in the systems, and clutch position tracking will be the goal of the controller derived in this paper.

Since pressurized air already is present in trucks, pneumatics are considered instead of hydraulics. Hydraulic actuators have been more common for position control of actuators, [2], [3], since pneumatics complicates the control task, due to the compressibility of air. But lately pneumatic have became more attractive due to decrease in component cost and improvement in valve technology [4], and several papers dealing with control of pneumatic actuators have been published the last years [4], [5]. On/off valves have been chosen over proportional valves for allocation of air because of cost and space advantages, and due to better robustness properties. The drawback with this choice is that these valves have a dynamic response which is hard to model accurately. Most commonly, such valves are controlled by Pulse Width Modulation (PWM) [6]-[8], but to take advantage of the switched control design, PWM is not applied in this paper. Instead we treat the valves as having pure on/off behavior, that is we assume that the valves can only be fully opened or fully closed. This leads to input restrictions which will restrict the control design.

Two switched controllers for this system have been developed in [9] and [10]. Both controllers show promising results, but also have some weaknesses. The one presented in [9] only ensure local stability results, while the controller in [10] ensures global stability, but is not as accurate as desired in the region around the equilibrium point. The main goal of this paper is therefore to combine these two switched controllers into one stable controller. We want to preserve the stability properties close to the equilibrium point of the local controller in [9] and the larger stability region of the global controller in [10].

II. System description

Figure 1 shows a sketch of the clutch actuator system. The Electronic Control Unit (ECU) calculates the control signals which are sent to the on/off valves. Based on these, the on/off valves control the flow to and from the actuator valve. The piston position is a result of the acting forces, the friction, the pressure and the spring forces, and this position decides the state of the clutch plates. The plates can either be engaged, disengaged or slipping. When engaged, the clutch transmits torque from the motor to the axle shaft.

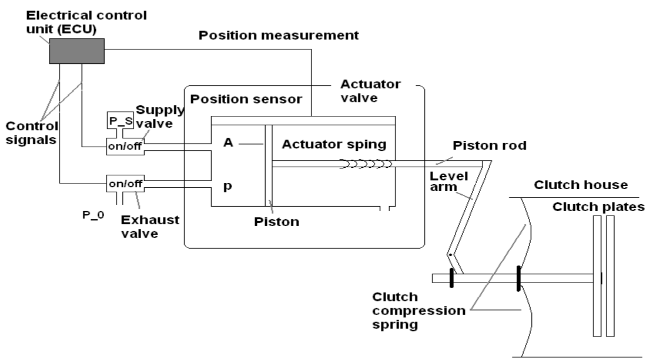

Fig. 1. Drawing of the electropneumatic clutch actuator system

As mentioned above, we only allow the on/off valves to be in one of the two positions, fully opened or fully closed. We have one valve for control of flow to and one for control of flow from the actuator valve, and both valves are never allowed to be fully opened at the same time, as their

*{hegesan,torj}@itk.ntnu.no, Department of Engineering Cybernetics, NTNU, Trondheim, Norway.

†christian.bratli@ka-group.com, Kongsberg Automotive ASA, Kongsberg, Norway.

effects would cancel each other out. This gives the following possible input values

$$w \in \{-U_{\max}, 0, U_{\max}\} \tag{1}$$

where $U_{\max}$ is the maximum flow capacity. This capacity will change depending on, among other factors, the pressure difference over the valve and the ambient temperature.

The system can be described by the error model

$$\begin{aligned} \dot{\tilde{x}}_1 &= x_2 \\ \dot{\tilde{x}}_2 &= \frac{1}{M}(-f_l + \frac{Ax_3}{Ax_1 + V_0} - AP_0 - Dx_2) \\ \dot{\tilde{x}}_3 &= RT_o w \end{aligned} \tag{2}$$

where x_1 is the position $[m]$ of the piston, x_2 is the velocity, $[m/s]$ and x_3 is the accumulated air in the actuator valve, $[kgm^2/s]$. The errors are described by $\tilde{x}_i = x_i - x_i^*$ and x_i^* are the reference points

$$x^* = [x_1^*, 0, x_3^*] \tag{3}$$

where x_1^* is given, and x_3^* is given by the equation

$$x_3^* = \frac{Ax_1^* + V_0}{A}(K_l(1 - e^{-L_l x_1^*}) - M_l x_1^* + AP_0)$$

The expression $f_l = K_l(1 - e^{-L_l x_1}) - M_l x_1$ describes the clutch load which is the combination of the actuator and the clutch compression springs. See the Appendix for definitions and values of variables.

This is a rather simple model of the system, suitable for control design, and the region of operation for this clutch actuator valve is $O = \{x \,|x_1 \in [0, 0.025], x_2 \in \mathbb{R}, x_3 \in [80, 504.68]\}$. The main uncertainty is the model of the clutch load, as this load changes with wear of the clutch and ambient temperature. This problem have been considered in [11] and further improvements are being pursued.

III. Controller design

A. Local controller

In [9] we developed the switched controller

$$w_1 = \begin{cases} -U_{\max} sgn(z_3), & z_3 \neq 0 \\ 0, & z_3 = 0 \end{cases} \tag{4}$$

where z are the following states found through backstepping

$$\begin{aligned} z_3 &= x_3 - \frac{Ax_1 + V_0}{A}(K_l(1 - e^{-L_l x_1}) \\ &\quad -M_l x_1 - Dkz_1 + AP_0 - Mkz_2) \\ z_2 &= x_2 + kz_1 \\ z_1 &= x_1 - x_1^* \end{aligned} \tag{5}$$

and k is a tuning parameter. With the Lyapunov function

$$V_1(z) = \frac{\alpha_1}{2} z_1^2 + \frac{\beta_1}{2} z_2^2 + \frac{\lambda_1}{2} z_3^2 \tag{6}$$

local exponential stability of the equilibrium point of the system (2) with the control law w_1 can be proved. The region of attraction is the invariant set

$$\bar{\Omega}_1 = \{z | V_1(z) \leq \bar{\epsilon}\}, \tag{7}$$

where $\bar{\epsilon}$ is the largest value such that $\bar{\Omega}_1 \subseteq \Omega_1$ and

$$\Omega_1 = \{z \,||a(z)| \leq RT_0 U_{\max}\} \tag{8}$$

where

$$\begin{aligned} a(z) &= -bz_3 - \frac{\beta_1 A z_2}{\lambda_1 M(Ax_1 + V_0)} + (z_2 - kz_1) \\ &\quad (K_l(1 - e^{-L_l x_1}) - M_l x_1 - Dkz_1 + AP_0 - Mkz_2) \\ &\quad + \frac{Ax_1 + V_0}{A}(z_2 - kz_1)(K_l L_l e^{-L_l x_1} - M_l - Dk) \\ &\quad - \frac{Mk(Ax_1 + V_0)}{A}(\frac{Az_3}{M\,(Ax_1 + V_0)} - k^2 z_1 - \frac{D}{M} z_2) \end{aligned} \tag{9}$$

and b is a backstepping parameter. The parameter k has a direct influence on the controller, and should be tuned such that the position error has as large influence on the choice of the input value as possible. Simulation result in [9] show that this controller holds good robustness properties.

B. Global controller

In [10] we developed the switched controller

$$w_2 = \begin{cases} -U_{\max} sgn(s(\tilde{x}_1, x_3)), & s(\tilde{x}_1, x_3) \neq 0 \\ 0, & s(\tilde{x}_1, x_3) = 0 \end{cases} \tag{10}$$

where

$$s(\tilde{x}_1, x_3) = \lambda_2 \tilde{x}_3 - \frac{\alpha_2}{M} \ln(\frac{Ax_1 + V_0}{Ax_1^* + V_0}) \tag{11}$$

For the region of operation, O, we have Lyapunov function

$$V_2(x) = \alpha_2 \int_0^{x_{1e}} -f(y, x_3)dy + \frac{\beta_2}{2} x_2^2 + \frac{\lambda_2}{2} \tilde{x}_3^2 \tag{12}$$

where

$$\begin{aligned} f(x_1, x_3) &= \frac{1}{M}(-K_l(1 - e^{-L_l x_1}) + M_l x_1 \\ &\quad + \frac{Ax_3}{Ax_1 + V_0} - AP_0). \end{aligned} \tag{13}$$

With this, asymptotic stability of the equilibrium point of the system (2) with the control law w_2, can be proved as long as $\frac{\alpha_2}{\lambda_2} \leq 3.839$. This restriction on $\frac{\alpha_2}{\lambda_2}$ makes it impossible to weight the position error as much as could be desired. Accumulated air has a magnitude which is 10^4 higher than the magnitude of the position, and the error in accumulated air becomes more important than the position error in the input decision when operating close to the reference point. This leads to poor robustness properties for this controller, as shown in [10], but it still works well far from the reference point.

C. Combined controller

We can use the global controller to bring the system close to the equilibrium point, and then apply the local controller, which depend more on the position error and has better robustness properties, close to the equilibrium point. We propose a combined controller

$$w_c = \begin{cases} w_2, & x \in \Omega_2 \\ w_1, & x \in \bar{\Omega}_1 \end{cases} \tag{14}$$

where the local controller is used if the system is inside the region, $\bar{\Omega}_1$, and the global controller is used elsewhere, in $\Omega_2 = O \backslash \bar{\Omega}_1$

Proposition 1: The equilibrium point of the system (2) with the switched controller given in (14) is asymptotical stable in the region O.

Proof: In the region Ω_2, $w_c = w_2$ guarantees asymptotic stability of the systems equilibrium point for the region O. In the region $\bar{\Omega}_1$, $w_c = w_1$ guarantees local exponential stability of the systems equilibrium point in the the region $\bar{\Omega}_1$. The equilibrium point lies inside $\bar{\Omega}_1$, and if the system at t_0 is in the region $\bar{\Omega}_1$ the exponential stability of the local controller will ensure that the region is not left and that the equilibrium point is reached. If the system at t_0 is in the region Ω_2 the asymptotical stability of the global controller ensures that the system will reach the region $\bar{\Omega}_1$ after some time t, and the equilibrium point will be reached with the local controller. ■

IV. Experimental results

A. Practical considerations

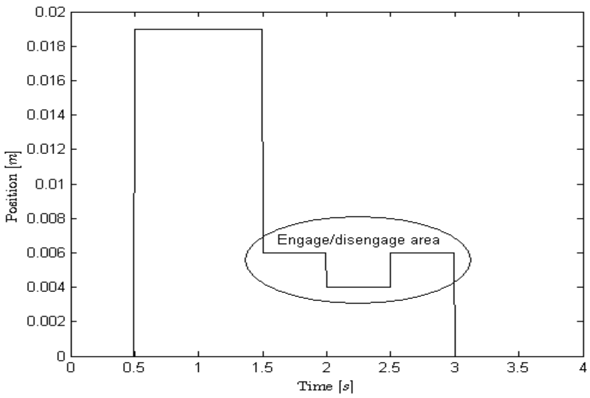

Fig. 2. Clutch sequence

A typical clutch sequence is shown in Figure 2. It is desired that the controller makes the system reach the reference point within $0.1\ s$, and with a steady state position error of less than $0.2\ mm$ in the area where the clutch engage/disengage. Outside this area, the requirements can be somewhat relaxed. The sampling time, and hence the switching time, is set to $1\ ms$. This is the same as the sampling rate of the position measurement and enough time for the on/off valves to open/close. The on/off valves also have a time delay of approximately $1\ ms$. To avoid unnecessary chattering, $w = 0$ have been chosen either whenever the value of the respective Lyapunov function, V_1 or V_2, is close to zero, which is the same as saying that x is close to x^*, or than the position error is close to zero. We call this Lyapunov and position error based deactivation of the controller.

Experiments have been conducted in a Scania test truck, shown in 3(a), at Kongsberg Automotive ASA, where the control algorithm run on the dSpace MABX 1401 unit shown in picture 3(b). Position and pressure are measured, while

Fig. 3. (a) The test truck (b) The setup with the dSpace unit

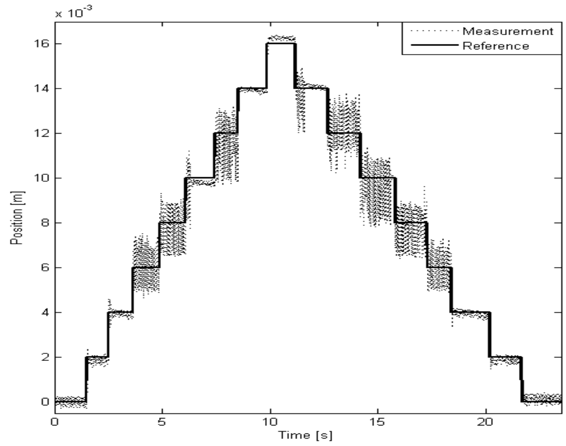

Fig. 4. Experiment for tuning of clutch load

velocity is calculated from the position measurement through a second order filter with coincident poles in $-40\ rad/s$, and accumulated air is calculated from

$$x_3 = pV(y). \tag{15}$$

By assuming $v = 0$ and $\dot{v} = 0$ we have

$$f_l = A(p - P_0) \tag{16}$$

from the motion dynamics. The parameters of the clutch load characteristic were tuned from the experiment shown in Figure 4 using the assumptions above. The parameters were found as

$$\begin{aligned} K_l &= 4000 \\ M_l &= 20000 \\ L_l &= 500 \end{aligned} \tag{17}$$

and Figure 5 show the resulting clutch load compared to experimental results obtained using (16).

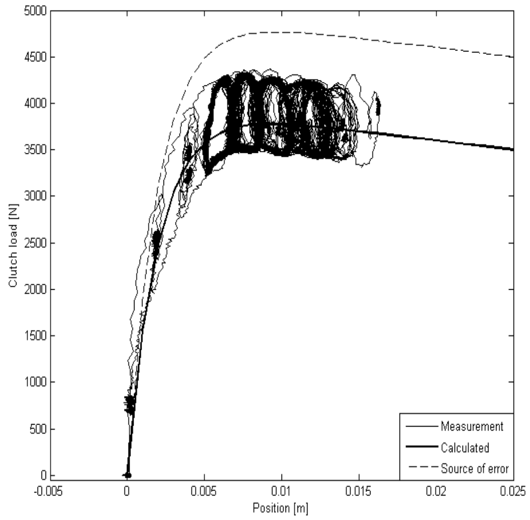

Fig. 5. Clutch load characteristic

B. Results

The performance of the combined controller is shown in Figure 6 and 7, where results from two experiments are shown. The effect of the deactivation of the controller is clear if comparing these results. In the experiment shown in Figure 6 a Lyapunov based deactivation which applies $w = 0$ if $V_1 \leq 300$ or $V_2 \leq 20$ is used, while in the experiments shown in Figure 7 a position error based de-activation which applies $w = 0$ if $|x_{1e}| \leq 0.4 \cdot 10^{-4}$ is used. The results obtained by Lyapunov based deactivation show little chattering of the input signal, but the accuracy in position is not as good as wanted, especially in the region of slipping. In the results obtained by position error based deactivation the accuracy in position is much better, but the chattering of the input signal is a bigger problem. A closer look at the position errors, displayed in Figure 8, gives us that the performance of the experiment with Lyapunov based deactivation not are acceptable with steady-state position errors up to 0.6 mm, while the position error based deactivation experiment perform due to the requirements if not for the chattering. When to deactivate the controller is consequently a trade off between chattering and accuracy. Chattering may be a big problem, as it causes the on/off valves to open and close often, and this can lead to overheated valve electronics.

The clutch load characteristic has a large influence on the system, and the characteristic will change during use due to change in temperature and wear of the clutch. We illustrate the impact by using another clutch load characteristic than the one found in (18) as a source of error in the controller, see Figure 5. Performance of all three controllers with this model error introduced are shown in Figure 9 and Figure 10. We see the expected behaviour, the global controller is more influenced by this error in the clutch load than the local controller. The local controller is, as stated above,

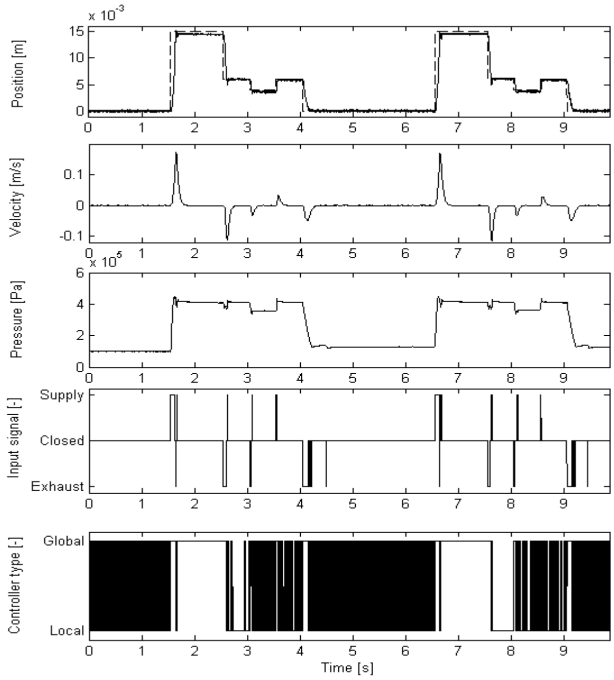

Fig. 6. Experiment with the combined controller, deactivation decided by the Lyapunov value

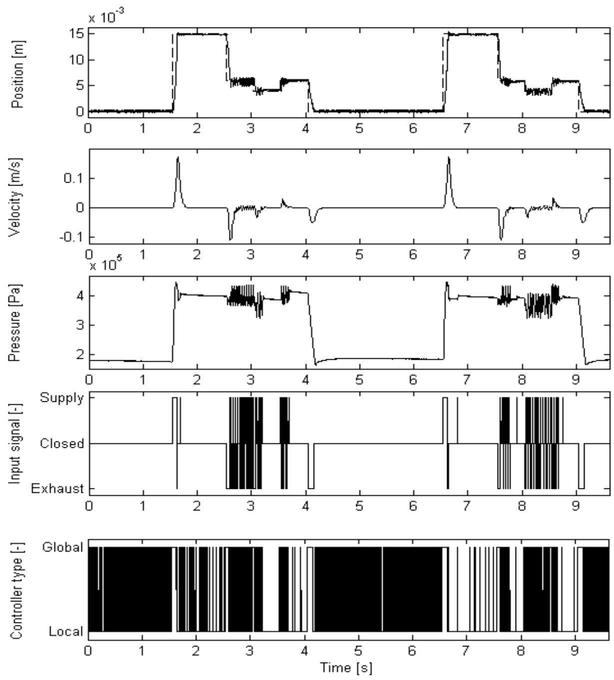

Fig. 7. Experiment with the combined controller, deactivation decided by position error

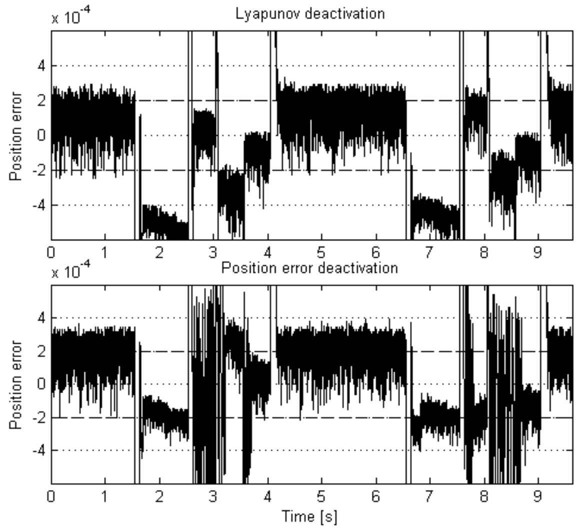

Fig. 8. Position error for the two experiments shown above

more robust and performs quit well even in the presence of this large model error. The global controller gives large position errors since the controller place emphasis on the error in accumulated air, and with model error, a small error in accumulated air no longer ensure a small position error. The combined controller, which uses the global controller until the stability region of the local controller is reached performs as expected, better than the global, but worse than the local controller. This sensitivity to errors in the clutch load motivates estimation of this characteristic [11].

The estimated region of attraction of the local controller might be too restrictive, as from simulations and experiments it seems like the controller guarantees stable behavior of the system for a larger region. It could help the robustness properties of the combined controller to increase the region Ω_1 to $|a(z)| < RT_0U_{\max}\delta$ where $\delta > 1$. But, one should have in mind that the local controller has no integrator term, and will not suppress drifting of the position error. These aspects, and the trade-off between accuracy and chattering following the deactivation of the controller, are two aspects which should be further investigated.

V. Concluding remarks

A stabilizing dual-mode switching controller for an electropneumatic clutch actuator have been derived. This controller is designed by combining two existing controllers for the same system, and preserving the best properties of the two individual controllers. Experimental results conducted in a truck, verifies the stability properties of the controller, and show that it is suited for the system. Further tuning of the controller is expected to improve the results, and is currently being pursued. The clutch load characteristic is the factor which influence the results the most, and the control design would benefit greatly from an online estimation [11].

Acknowledgments: This work has been sponsored by the Norwegian Research Council and Kongsberg Automotive ASA.

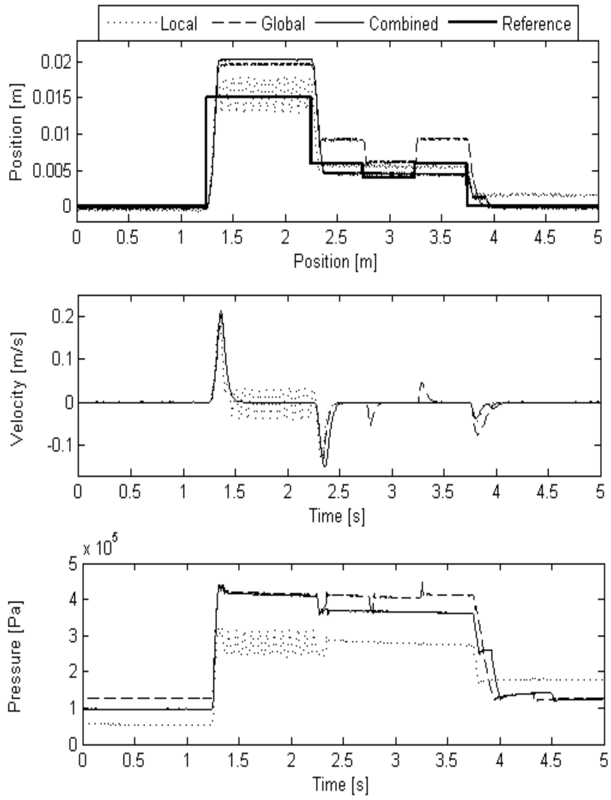

Fig. 9. Performance of the controllers with introduced model error

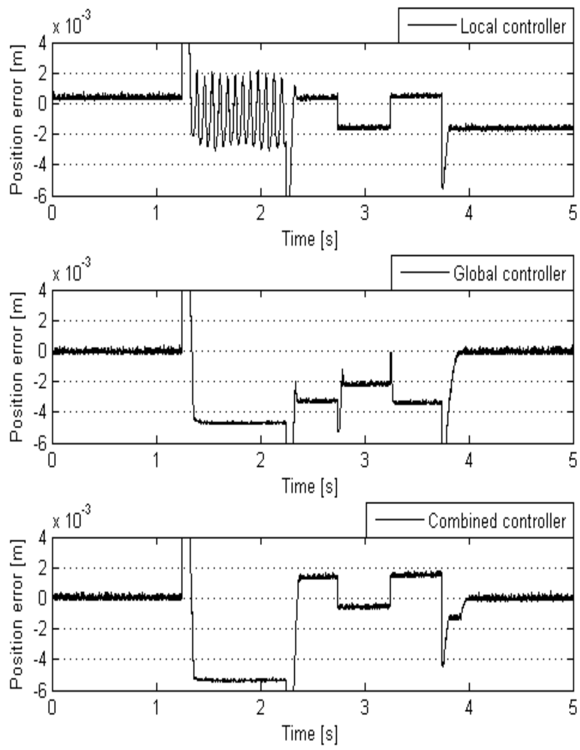

Fig. 10. The position errors of the experiments shown in Figure 9

REFERENCES

[1] G. O. Kaasa, "Nonlinear output-feedback control applied to electro-pneumatic clutch actuation in heavy-duty trucks," Ph.D. dissertation, NTNU, 2006.

[2] J. Zhang, L. Chen, and G. Xi, "System dynamic modelling and adaptive optimal control for automatic clutch engagement of vehicles," in *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, vol. 216, 2002, pp. 983–991.

[3] G. Lucente, M. Montanari, and C. Rossi, "Modelling of an automated manual transmission system," *Mechatronics 17 (2007)*, pp. 73–91, 2007,

[4] T. Nguyen, J. Leavitt, F. Jabbari, and J. E. Bobrow, "Accurate sliding-mode control of pneumatic systems using low-cost solenoid valves," *IEEE/ASME Transactions on mechatronics*, vol. 12, pp. 216–219, 2007,

[5] K. Ahn and S. Yokota, "Intelligent switching control of pneumatic actuator using on/off solenoid valves," *Mechatronics*, vol. 15, pp. 683–702, 2005,

[6] R. B. V. Varseveld and G. M. Bone, "Accurate position control of a pneumatic actuator using on/off solenoid valves," *IEEE/ASME Transactions on mechatronics*, vol. 2, pp. 195–204, 1997,

[7] K. Peterson, J. W. Grizzle, and A. G. Stefanopoulou, "Nonlinear levitation of automative engine valves," 2004,

[8] Y. Wang, A. Stefanopoulou, M. Haghgooie, I. Kolmanovsky, and M. Hammoud, "Modeling of an electromechanical valve actuator for a camless engine," in *Proceedings AVEC 2000, 5th Int'l Symposium on Advanced Vehicle Control*, vol. 93, 2000,

[9] H. Sande, T. A. Johansen, G. O. Kaasa, S. R. Snare, and C. Bratli, "Switched backstepping control of an electropneumatic clutch actuator using on/off valves," in *Proceedings of the 26th American Control Conference, New York*, 2007, pp. 76–81.

[10] H. Langjord, T. A. Johansen, and J. P. Hespanha, "Switched control of an electropneumatic clutch actuator using on/off valves," in *Proceedings of the 27th American Control Conference, Seattle*, 2008.

[11] H. Langjord, T. A. Johansen, S. R. Snare, and C. Bratli, "Estimation of electropneumatic clutch actuator load characteristics," in *Proceedings of the 17th IFAC World Congress, Seoul*, 2008.

APPENDIX

Variable	**Value**	**Unit**	**Description**
A	$12.3 \cdot 10^{-3}$	m^2	Actuator area
P_0	$1.095 \cdot 10^5$	Pa	Ambient pressure
T_0	293	K	Temperature
R	288	$\frac{J}{kgK}$	Gas constant of air
M	10	kg	Mass of piston
P_S	$9.5 \cdot 10^5$	Pa	Supply pressure
D	2000	$\frac{Ns}{m}$	Viscous damping
K_l	4000	N	Load char. term
L_l	500	$-$	Load char. term
M_l	20000	$\frac{N}{m}$	Load char. term
V_0	$0.148 \cdot 10^{-3}$	m^3	Volume at $y = 0$
$U_{\max}$	0.003	$\frac{kg}{s}$	Maximum flow
k	$382/10$	$-$	Backstepping parameter
b	10	$-$	Backstepping parameter
α_1	400	$-$	Tuning parameter
β_1	$0.0027/4$	$-$	Tuning parameter
λ_1	0.01	$-$	Tuning parameter
α_2	3.839	$-$	Tuning parameter
β_2	3.839	$-$	Tuning parameter
λ_2	1	$-$	Tuning parameter